Best vibration monitoring practice for large ABB gas turbine protection and machine management

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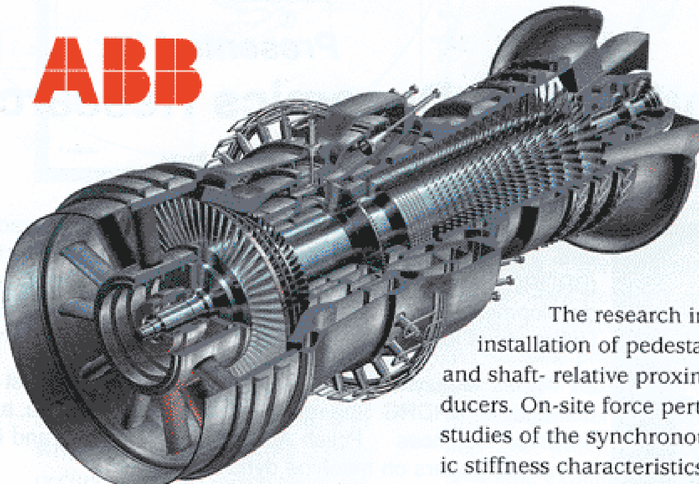

Deregulation of the power industry has imposed a new set of demands on gas turbine manufacturers. Generating companies are demanding higher efficiencies and improved reliability, state and local governments are demanding that new and existing machines meet stringent emission standards, and power customers are demanding lower energy costs. In an effort to satisfy these demands, gas turbine manufacturers, such as ABB, have designed large combined cycle power trains that have demonstrated efficiencies above 57%, NOx emission levels below 25 ppm and a dollars/kWh rate that is over 30% lower than previous gas turbine designs.

Due to this demand, and the difficulty associated with testing large machines, there were problems encountered by all of the leading gas turbine manufacturers as this new class of machines was introduced. Recognizing a need to protect the units and to provide users with adequate tools for managing the machines and ensuring their reliability, ABB approached Bently Nevada and requested assistance with developing a measurement system that would provide increased value to their customers; that is, a "Best Practice System."

Historically, ABB has used seismic transducers on bearing pedestals as the primary source of vibration information from their large industrial gas turbines. However, Bently Nevada Corporation strongly promotes and advocates the use of proximity probe shaft vibration transducers on all machines equipped with fluid film bearings (including large industrial gas turbines). Significant research has been conducted jointly by ABB and Bently Nevada for the last three years to determine the selection of vibration transducers that are used for protecting, monitoring, and managing these machines.

The research included installation of pedestal seismic and shaft- relative proximity transducers. On-site force perturbation studies of the synchronous dynamic stiffness characteristics of ABB's GT rotor and bearing pedestals were performed, as well as development of a comprehensive rotor dynamic model to assess the dynamic behavior of this particular machine. Studies of both shaft and pedestal vibration, resulting from various perturbation forces and frequencies, were performed. Direct input of field data into the rotor dynamic model has provided actual bearing and pedestal dynamic stiffness values. The tests and the conclusions are described in references [1] & [2]. This engineering study conducted by ABB and BNC resulted in the development of a Best Practice System.

Best Practice for large gas turbine machinery from ABB

For protection against machine malfunctions that result in excessive pedestal vibration, the XY

pedestal seismic vibration measurement is *more sensitive to undesired vibration* and is used for vibration protection (alarm and shutdown). Alarm annunciation *may* also be provided on the XY shaft relative vibration measurement for machine management.

In cases of stiff bearing supports, for example, the generator, where the amplitude of the shaft relative vibration exceeds the amplitude of the pedestal vibration, the relative vibration measurement is more sensitive to undesired vibration and is used for vibration protection (alarm and shutdown). Alarm annunciation may also be provided on the bearing support vibration measurement for machine management.

There are a number of machinery characteristics that can only be detected when both pedestal and shaft vibration transducers are provided as a part of the machinery protection and management system. If XY shaft relative vibration measurements are not provided, these machine characteristics cannot be effectively detected and assessed. Therefore, for diagnostic purposes, the use of both bearing cap and shaft relative vibration measurements is beneficial.

The question is, what is the Best Practice for this large gas turbine application?

The recommendation for this machine design is that seismic transducers mounted to the bearing cap offer *protection*. However, for *complete machinery management*, both shaft relative and casing seismic measurements should be installed. This will allow for the effective detection and diagnosis of the majority of machine vibration malfunctions and provide for proactive management of these turbines.

Experiences from Operation and Supporting Evidence

ABB's large industrial gas turbines are, in general, supported on non-rigid bearing pedestals with moderate dynamic stiffness. This results in pedestal seismic vibration that *normally* exceeds the shaft relative vibration.

A typical machine train layout of an ABB gas turbine application is shown in Figure 1. When the signals from a seismic transducer at the #1 bearing cap and a proximity probe at the same location are compared during a typical startup (Figure 2), the seismic measurement shows higher amplitudes through the two resonance frequencies than the shaft relative measurement. This is due to the compliance of the gas turbine support, which has been designed to move with the rotor during an unbalance condition. The phase measurements of the two signals reveal that they track each other very closely during the entire startup, and cross at the 2350 rpm resonance. This type of response indicates a machine pedestal with relatively moderate dynamic stiffness and confirms that casing mounted seismic transducers can be used to effectively protect and balance the gas turbine.

When an alarm occurs due to an increase in bearing vibration, it is

Brg .#	Description	Brg .#	Description
1	Turbine Bearing	3	Generator Bearing drive end
2	Compressor Bearing	4	Generator Bearing non-drive end
T	Thrust Bearing	5	Exciter Bearing

Figure 1. Typical ABB GT train configuration.

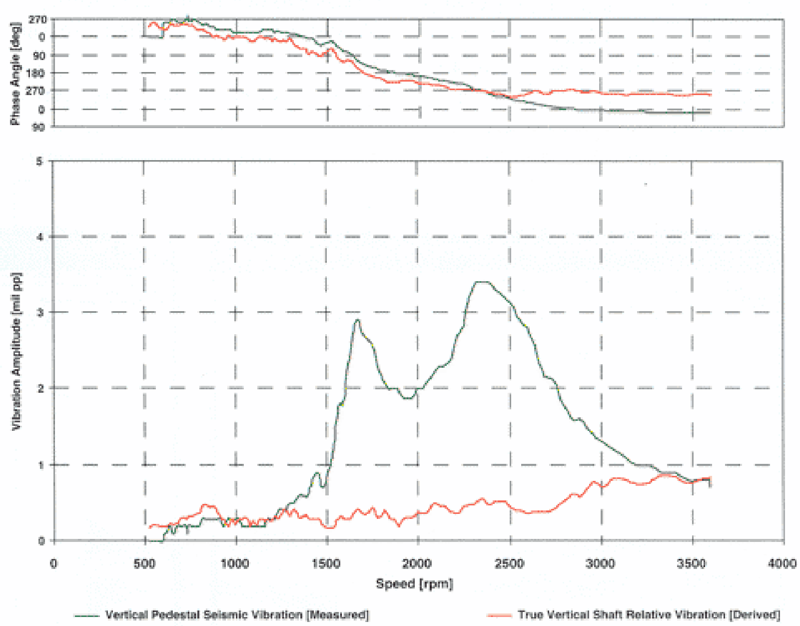

Figure 2. GT startup Bode plot of compressor - end bearing. Bearing cap seismic data (green) and shaft relative displacement data (red).

important for operators to have access to additional information that allows them to quickly and correctly respond to the alarm condition. Many vibration characteristics associated with increased vibration levels can only be observed by using shaft proximity transducers. Some of these characteristics are:

- Shaft average centerline position
- Shaft orbital motion
- Direction of shaft orbital precession

Shaft orbit and centerline plots reveal misalignment

While reviewing the plots from the force perturbation studies, a comparison of the shaft orbit plots from the XY proximity transducers installed in bearing #2 (the compressor bearing, Figure 3) and bearing #3 (the generator drive end bearing, Figure 4) indicated that a fairly large load existed at bearing #2. Due to the asymmetry between horizontal and vertical dynamic stiffness properties of the bearing pedestal and the increased load, the shaft orbit plot should be forward in precession and elliptical, with the major axis of the ellipse within 45° of the horizontal plane. The orbit plot for bearing #2 was rather small, highly elliptical, and forward in precession, while the orbit plot for bearing #3 was more than twice as large, more circular, and also forward in precession. (Precession refers to the direction of the shaft orbital motion. If the shaft orbital motion is in the same direction as shaft rotation, the vibration has forward precession. If the shaft orbital motion is opposite to shaft rotation, the vibration has reverse precession.) As the shaft becomes more highly loaded in the bearing, the orbital response becomes more elliptical, progressing to a straight line or "C" shaped orbit, which contains reverse precession components.

Although most machines can operate for an extended period of time with increased load, this is undesirable due to the load reversal stresses that are associated with the highly elliptical or reverse precession orbit experienced by the shaft and couplings and the thermal and mechanical stresses experienced by the bearing. Because of this, ABB, BNC, and the owner of the gas turbine decided to initiate a root cause analysis to determine the severity of, and the reason for, the increased load.

The shaft centerline plots from bearings #2 and #3 were reviewed (Figures 5 and 6). The plot for bearing #2 revealed that the turbine rotor remained in the lower portion of the bearing as the machine warmed up. The plot for bearing #3 revealed that the generator rotor started at the bottom of the bearing and moved toward the center of the bearing during this same time period. This explained the constrained orbit plot at bearing #2 (Figure 3): the rotor was being forced into the bottom of the bearing, with a resultant increase in dynamic stiffness, by the increased load it was assuming from bearing #3. It also explained why the orbit plot at bearing #3 (Figure 4) was more circular: the dynamic stiffness decreased as the rotor was lifted away from the bearing wall towards the center. (The eccentricity ratio at bearing #2 was close to one while the eccentricity ratio at bearing #3 was approaching zero.) This type of response is indicative of misalignment, and an alignment study was recommended to verify proper alignment between the gas turbine and generator.

The alignment study revealed that, when the machine cases were cold, the generator rotor was aligned below the turbine rotor (Figure 7). Therefore, under cold conditions, the generator needed to be raised. Also, under hot condi-

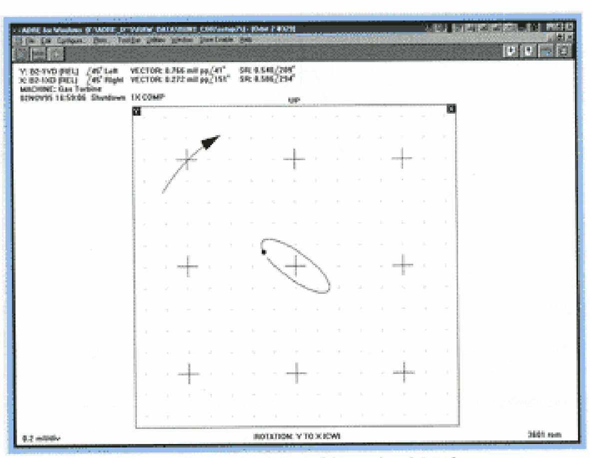

Figure 3. Bearing #2 1X filtered orbit plot.

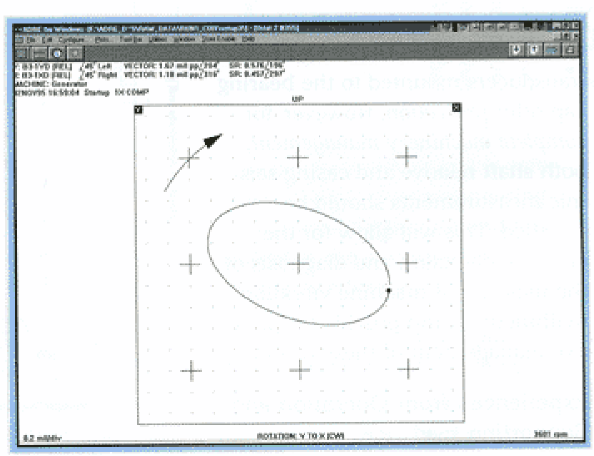

Figure 4. Bearing #3 1X filtered orbit plot.

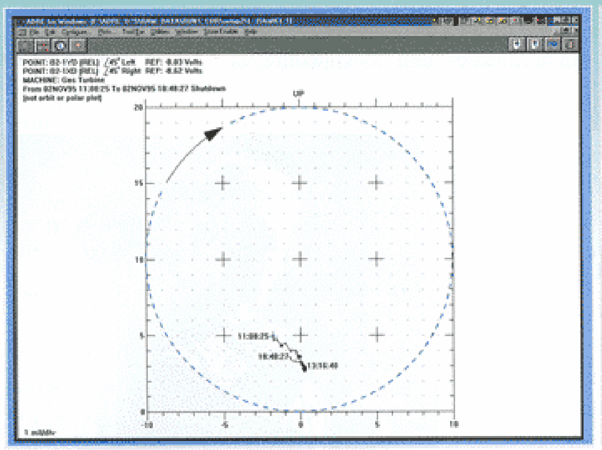

Figure 5. Bearing #2 shaft centerline plot.

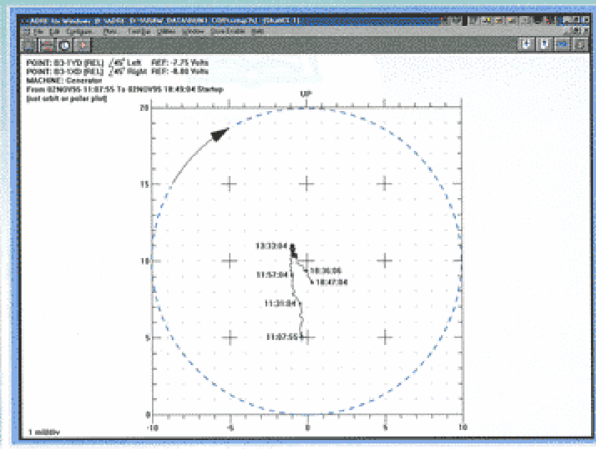

Figure 6. Bearing #3 shaft centerline plot.

tions, the compressor end of the GT turbine grows more in the vertical direction than the hot turbine end. Both effects lead to an increased loading of the compressor bearing (bearing #2). This misalignment also helped to explain the increase in bearing metal temperatures observed at bearing #2 after the machine ran for a few hours.

Because of the type of misalignment, the seismic transducers did not measure any significant change in machinery characteristics and could provide no information concerning the amount of misalignment. This application also demonstrates the usefulness of shaft relative proximity probes, for rotor vibration and position data (that is, probe gap data), to provide information for root cause analysis.

Conclusion

The design of a vibration monitoring system for a unit has to meet two major requirements. First, machine protection has to assure a safe operation of the unit, and second, the announcement of an alarm should include enough information for analysis of the root cause and recommendation of corrective action. ABB and BNC recommend the installation of a system which combines seismic transducers for machine protection and proximity transducers for machinery management and improved reliability. The Best Practice system described in this article meets these requirements. The alignment case described earlier shows that this system provides enough information for an appropriate evaluation.

Due to the knowledge obtained during the force perturbation studies, ABB and BNC feel very confident that the system described as the Best Practice system will provide maximum value to the gas turbine user. This knowledge would not have been obtained without the assistance of Bently Nevada's Machinery Management Services department, the Bently Rotor Dynamics Research Corporation and the support of Kentucky Utilities, who instrumented and operated one gas turbine especially for this research effort. Both ABB and BNC thank these organizations for their significant contributions.

References

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2. Florjancic, S., Lively, N., Thomas, R., Mechanical Behavior of an Industrial Gas Turbine under Fault Conditions, a Case History.

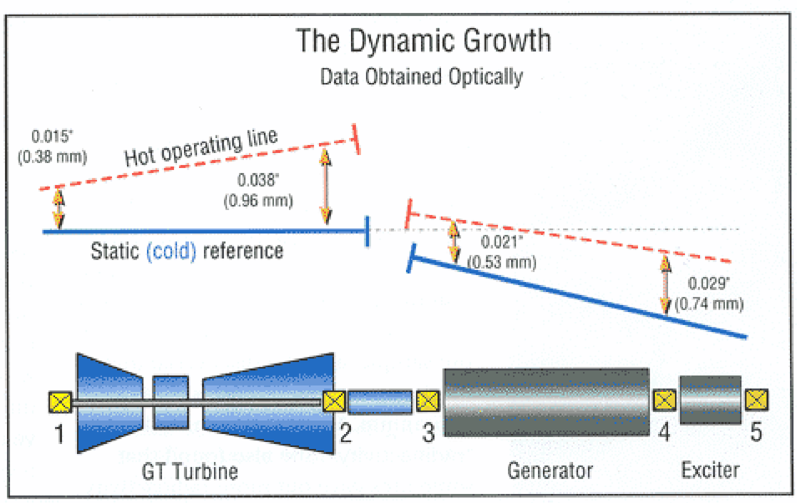

Figure 7. Results of vertical alignment study